OPERATING STABILITY OF THE APOLLO FUEL-CELL CONDENSER

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SUMMARY

An experimental evaluation of the Apollo fuel-cell condenser conducted at the Lewis Research Center showed that there are no operating instabilities in the small condensing tubes.

Motion pictures taken of the condensate emerging from the condensing passages coupled with traces of the fluctuations in the pressure differential across the condenser show that the condensate flow in the tubes is controlled by surface tension and viscous forces, and that operation in zero gravity should not alter the condenser's operation.

INTRODUCTION

The Apollo fuel-cell condenser is intended to remove both the water and waste heat produced by the fuel cell. The condenser is to operate over a wide range of inlet conditions in a zero-gravity environment.

In order to determine if there were any condensing instabilities in the small condensing tubes, or if the mode of condensation was such that instabilities might occur in zero-gravity operation, a ground test evaluation was conducted at the Lewis Research Center.

The condenser removes the product water by condensing it from a recirculating hydrogen and water-vapor stream. It transfers the excess heat from this gas stream to a glycol-water solution which in turn passes through a radiator.

Photographs of the condenser (figs. 1 to 3) show the small trapezoidal flow passages in which the condensation takes place and the placement of the coolant and gas stream inlet and outlet passages. Drawings of the condenser (figs. 4 and 5) show the internal placement of the flow passages. The condenser has been designed to condense 0.8 pound per hour of water under the following conditions:

E-2686
Hydrogen flow, lb/hr .................................. 5.0
Water vapor flow, lb/hr ................................ 11.4
Condensate, lb/hr ...................................... 0.8
Coolant flow, lb/hr ..................................... 80
Gas inlet temperature, °F ................................ 253
Gas outlet temperature, °F ................................ 200
Coolant inlet temperature, °F ............................ 181
Coolant outlet temperature, °F ........................... 211
Operating pressure, psia ................................ 60

Under the most unfavorable conditions specified, the condensing rate was 2.34 pounds per hour. This condition occurs when the fuel cell is operating at maximum power output (2200 W).

SYMBOLS

A    orifice area, sq in.
C_w  orifice coefficient
H_s  specific humidity, (lb H_2O)/(lb H_2)
K    gas constant, (lb mass)(°R)/(lb force)(sec)
P    pressure upstream of orifice, psia
T    temperature, °R
t    time, sec
\dot{w}  gas flow rate, lb/hr
Y    coolant flow rate, lb/hr

APPARATUS AND PROCEDURE

A test rig designed for extensive control of all important condenser operating parameters (inlet temperatures, gas and coolant flow rates, and operating pressures) was assembled (fig. 6). Special gas inlet and outlet headers were fabricated for the condenser to facilitate the measurement of pertinent gas parameters while allowing the photographing of the two-phase flow at the condenser exit (figs. 7 and 8).

The parameters under operator control were the following:
The measured parameters were the following:

1. Gas outlet temperature
2. Water condensation rate
3. Pressure differential $\Delta P$ across the condensing tubes
4. Coolant outlet temperature

The manner in which these parameters were measured and controlled will now be discussed.

The flow rates of the hydrogen and water vapor streams were controlled by controlling the temperatures and pressures at the respective choked orifices. These orifices (fig. 9) were calibrated (flow rate as a function of upstream pressure and temperature) by using either nitrogen or helium. From these calibrations the orifice areas were calculated from the choked orifice equation

$$
\dot{w} = \frac{3600 \text{ PAC}_w K}{T^{1/2}}
$$

where $C_w$ was assumed equal to 0.83. The flow rates of hydrogen and water vapor were calculated from these areas and the respective constants ($K$) for each gas. The temperatures and pressures upstream of the orifices were measured with Chromel-Alumel thermocouples and Bourdon gages, respectively. In this manner the temperatures and pressures were read to precisions of $\pm 1^\circ$ F and $\pm 0.5$ psia.

The operating pressures were measured and recorded as follows: The gas pressure at the condenser outlet header was measured with a 10-inch, 100-psig Heise gage and also was recorded on an XYXY' plotter by using the electrical output from a pressure transducer.

The pressure differential $\Delta P$ across the condensing tubes was measured with a 0.15-psid transducer that was calibrated at least once a week and zero checked prior to every run.

All temperatures were measured with Chromel-Alumel thermocouples referenced to $32^\circ$ F and read from a digital voltmeter.

Some difficulty was encountered in obtaining accurate readings of the condenser outlet temperature. The water ejected from the condensing tubes was found to be striking the thermocouple probe (fig. 7(a)) and causing lower than expected readings. To im-
prove this situation a liquid drop deflector made of plastic tubing was placed over the thermocouple.

The flow rate of the glycol-water solution used as the coolant was batch measured by using a 1000-milliliter volumetric flask and a stopwatch. The time periods needed to fill the flask, measured to 0.1 second, were used to calculate the flow rate.

**DATA REDUCTION - ERROR ANALYSIS**

The calculations made when using the experimental data were as follows:

1. Hydrogen and steam flow rates through the choked orifices
2. Inlet humidity to the condenser
3. Coolant flow rates

**Gas Flow Rates**

Since there are errors inherent in the method used to measure or calculate the values of all the terms in the choked orifice flow equation

\[
\dot{w} = \frac{3600 \text{ PAC}_w K}{T^{1/2}}
\]  

(1)

there are possible errors in the calculated values of \( \dot{w} \). A maximum value for the error in \( \dot{w} \) can be found by differentiating equation (1) and adding all terms:

\[
\begin{align*}
\frac{d\dot{w}}{\dot{w}} &= \frac{3600 \text{ PAC}^*_w dK}{T^{1/2}} + \frac{3600 \text{ PAC}_w dC_w}{T^{1/2}} + \frac{3600 \text{ PC}_w^* K dA}{T^{1/2}} \\
&\quad + \frac{3600 \text{ AC}^*_w dP}{T^{1/2}} + \frac{3600 \frac{1}{2} \text{ PAC}_w dT}{T^{3/2}}
\end{align*}
\]

(2)

where \( d\dot{w} \) is related to the errors in all the other quantities (\( dP \), \( dT \), etc.).

To obtain an idea of the magnitudes of \( d\dot{w} \) for both steam and hydrogen, sample data points are utilized as follows:
For steam:

\[
\begin{align*}
P &= 142 \text{ psia} \quad \text{d}P = \pm 0.5 \text{ psia} \\
T &= 809^\circ \text{ R} \quad \text{d}T = \pm 2^\circ \text{ R} \\
C_w &= 0.83 \quad \text{d}C_w = 0 \quad (C_w = 0.83) \\
A &= 1.95 \times 10^{-3} \text{ sq in.} \quad \text{d}A = 1 \times 10^{-5} \text{ sq in.} \\
K &= 0.412 \quad \text{d}K = 1 \times 10^{-3}
\end{align*}
\]

When direct substitution is made in equations (1) and (2),

\[
\frac{\Delta w}{w} = 0.16 \text{ lb/hr}
\]

\[w = 12.0 \text{ lb/hr}\]

and

\[100 \frac{\Delta w}{w} = 1.3 \text{ percent maximum}\]

For hydrogen:

\[
\begin{align*}
P &= 168 \text{ psia} \quad \text{d}P = \pm 0.5 \text{ psia} \\
T &= 501^\circ \text{ R} \quad \text{d}T = \pm 2^\circ \text{ R} \\
A &= 1.91 \times 10^{-3} \text{ sq in.} \quad \text{d}A = 1 \times 10^{-5} \text{ sq in.} \\
K &= 0.1402 \quad \text{d}K = 0.2 \times 10^{-3} \\
C_w &= 0.83 \quad \text{d}C_w = 0
\end{align*}
\]

When direct substitution is made with hydrogen data values,

\[100 \frac{\Delta w}{w} = 1.3 \text{ percent maximum}\]

The condenser inlet humidity

\[
H_s = \frac{\dot{w}_{H_2O}}{\dot{w}_{H_2}} \quad (3)
\]
when differentiated is

$$dH_s = \frac{\dot{w}_{H_2}(\dot{w}_{H_2} - \dot{w}_{H_2}O)}{(\dot{w}_{H_2})^2}$$

Direct substitution of the previous flow rates yields

$$dH_s = 0.07$$

$$H_s = 2.0$$

and

$$\frac{100}{H_s} dH_s = 4\text{ percent maximum}$$

For all data points the following are assumed:

$$\frac{\dot{w}_{H_2}O}{\dot{w}_{H_2}} = 1.3\text{ percent}$$

$$\frac{\dot{w}_{H_2}}{\dot{w}_{H_2}} = 1.3\text{ percent}$$

$$\frac{dH_s}{H_s} = 4\text{ percent}$$

It is noted that all possible errors are maximized. The flow rate (lb/hr) of coolant is directly related to the time needed (in sec) to fill a 1000-milliliter volumetric flask. When a density of 1.026 grams per cubic centimeter is used for a solution of 70 weight percent glycol in water, \(^1\)

\(^1\)Extrapolated from data for a 62.5 percent solution at 200° F found in reference 1.
\[ Y(\text{lb/hr}) \text{ of coolant} = 0.123 \ Y(\text{cc/sec}) \]

\[ Y(\text{lb/hr}) \approx \frac{10000 \ \text{sec}}{1.23 \ Y \ 1000 \ \text{cc}} = t(\text{sec}) \]

\[ Y = \frac{8120}{t} \tag{4} \]

Now from equation (4),

\[ \frac{dY}{dt} = \frac{8120}{t^2} \]

for

\[ t = 100 \ \text{sec} \]

\[ dt = 0.1 \ \text{sec} \]

\[ dY = 0.08 \approx 0.1 \ \text{lb/hr} \]

Therefore a maximum error in \( Y \) of 0.2 pound per hour is assumed.

**RESULTS AND DISCUSSION**

Traces of the gage pressure at the condenser exit header and the \( \Delta P \) across the condenser as functions of time were recorded on an XYY' plotter (samples, fig. 10).

From these traces it is evident that the overall fluctuations in \( \Delta P \) are a small percentage of the full \( \Delta P \) values; however, individual tube fluctuations may be larger than those shown on the plots, since the taps for the \( \Delta P \) transducer are in the large volume headers at each end.

Another fact indicated by both the data (table I) and the \( \Delta P \) traces is that the average \( \Delta P \) is fairly constant (0.06 psi) over all the runs. The differences shown in table I (0.05 to 0.07 psi) do not seem to be related to any changes in operating conditions.

A number of tests at condensing rates above the Apollo mission maximum (2.34 lb/hr) showed that even under these conditions operation was stable (fig. 10(b), table I). Motion pictures of the condensate emerging from the condensing tubes at three condensate rates (0.96, 1.06, and 3.06 lb/hr) were taken at normal (24 fps) and high (240 fps) camera speeds.
<table>
<thead>
<tr>
<th>Date</th>
<th>Run time, min</th>
<th>Steam</th>
<th>Hydrogen</th>
<th>Humidity, lb/hr</th>
<th>Condenser</th>
<th>Flask fill time, sec</th>
<th>Coolant</th>
<th>Condensate removal rate, lb/hr</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Orifice pressure, psia</td>
<td>Orifice temperature, °R</td>
<td>Orifice area, sq in.</td>
<td>Orifice pressure, psia</td>
<td>Orifice temperature, °R</td>
<td>Orifice area, sq in.</td>
<td>Flow rate, lb/hr</td>
</tr>
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<td>996</td>
<td>1.95 x 10^-3</td>
<td>11.50 x 0.15</td>
<td>141</td>
<td>503</td>
<td>1.91 x 10^-3</td>
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<td>937</td>
<td>1.95 x 10^-3</td>
<td>11.30 x 0.15</td>
<td>140</td>
<td>492</td>
<td>1.91 x 10^-3</td>
</tr>
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<td>498</td>
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<td>870</td>
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<td>11.05 x 0.14</td>
<td>177</td>
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<td>13.10 x 0.17</td>
<td>173</td>
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<td>906</td>
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<td>524</td>
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<td>11.10 x 0.14</td>
<td>167</td>
<td>499</td>
<td>1.91</td>
</tr>
</tbody>
</table>

a1964.

bUncalibrated ΔP.

cPhotographs of run made; data inconsistent.
These movies show that the condensate does not "flow" out of the tubes as it is formed, but collects in each tube until it is ejected as a slug or a stream of drops. The movies show that this mode of operation is common to all of the 260 condensing tubes, and also that the only difference caused by an increase in condensing rate is that more slugs per unit time are ejected.

A sequence of frames taken from one of the films is reproduced in figure 11. These frames, covering a time period of 1/10 second, show a condensate slug emerging from a tube.

This condensing mode is consistent with the traces of $\Delta P$ against time. The movies show that at any given time some tubes are filling with condensate and a smaller number of tubes are ejecting. The ejecting process accounts for the momentary $\Delta P$ increases, and the averaging out of the condensate ejecting into the header from the 260 tubes accounts for the very stable mean $\Delta P$.

In order to show that surface tension forces, acting on the condensate in the tubes, do predominate over those of gravity when the condenser operates in a horizontal position, an experiment to measure the surface tension head in the tube was performed.

A degreased condenser was held in the vertical position over a water bath so that the tube exits were at the surface of the water. Then the gas pressure needed to force the water out of a tube was measured (fig. 12).

The distribution plate at the gas inlet end of the condenser was removed in order to apply gas pressure to single tubes. Gas pressure was built up by letting water drip slowly into a closed vessel, and the gas was introduced into single tubes through a hypodermic needle. The gas pressure was then read from an inclined water manometer when a bubble was seen emerging from a tube exit.

In this experiment, the gas pressures needed to blow the water out were measured for 45 tubes (table II). The average gas pressure was found to be 1.65 centimeters of water with most of the points falling between 1.55 and 1.75 centimeters (36 of 45).

This value of 1.65 centimeters is very close to that which can be calculated for a trapezoidal tube assuming perfect wetting (contact angle, $0^\circ$) and an effective radius:

$$ h = \frac{2\gamma}{rdg} $$

where

- $h$ head, cm of water
- $\gamma$ surface tension, dynes/cm
- $r$ effective radius of tube, $(A/\pi)^{1/2}$, cm
- $d$ density of water, gm/cc
- $g$ 980 cm/sec$^2$
TABLE II. - MEASURED VALUES OF GAS PRESSURE NEEDED TO FORCE LIQUID OUT OF A SINGLE TUBE

<table>
<thead>
<tr>
<th>Tube</th>
<th>Pressure, cm of water</th>
<th>Tube</th>
<th>Pressure, cm of water</th>
<th>Tube</th>
<th>Pressure, cm of water</th>
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<td>1.72</td>
<td>30</td>
<td>1.74</td>
<td>45</td>
<td>1.75</td>
</tr>
</tbody>
</table>

\(^{a}\)Average pressure, 1.65 centimeters of water.

The area of one tube is approximately \(2.34 \times 10^{-2}\) square centimeter (fig. 5), and the radius of a circle having this area is 0.062 centimeter. Consequently,

\[
h = 1.94 \text{ cm of water}
\]

when

\[
d = 1.00 \text{ gm/cc}\]
\[
g = 980 \text{ cm/sec}^2\]
\[
\gamma = 59 \text{ dynes/cm at } 200^\circ F
\]

Since one tube is only about 0.07 inch high (0.18 cm), gravitational forces would not be expected to have much of an effect on the condensate behavior in a tube.

On the basis of the data presented and a study of the condensing pictures, it can be stated that the surface tension and dynamic pressure differential forces acting on the condensate in the tubes are prime factors controlling condensate removal.

The liquid condensate is held in a tube by surface tension forces acting on the tube interior and at the tube exit. When sufficient pressure head is built up to overcome these forces the slug is blown free.
Since the controlling force during one-gravity operation (tubes horizontal) is surface tension, rather than gravity, it is evident that under zero-gravity conditions the surface tension effect will be predominant.

CONCLUSIONS

1. With the condenser operating horizontally in a one-gravity environment, there is no evidence of any operating instability over a wide range of condensate rates.
2. Small differences in the tube to tube condensing modes are averaged out by the large number of tubes to give very stable operation.
3. Since the surface tension rather than gravity forces control the condensate removal mechanism, no problem areas are envisioned in zero-gravity operation.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, February 12, 1965.

REFERENCE

Figure 1. - Apollo fuel-cell condenser.

Figure 2. - Condenser exit.
Figure 3. - Condenser gas inlet.

Figure 4. - Condensing and coolant passages.
Section of core (x10)

Coolant passage, 9 places alternating with reactant passages

Figure 5. - Condenser. (All dimensions in inches.)
Figure 6. - Flow diagram for condenser experiment.
Figure 7. - Exit header.

(a) Back view.

(b) Top view.

- Pressure taps
- Gas exit line
- Covered thermocouple
- View window
- Condensate drain
Figure 8. - Hydrogen plus water condenser assembly. (All dimensions in inches.)
Figure 9. - Flow control orifice.

(a-1) Total pressure, 45.0 psia.

(a-2) Pressure differential, 0.057 psia.
(a) Date, 2/6/64; condensing rate, 1.1 pounds per hour.

(b-1) Total pressure, 45.0 psig; reading from Heise gage.
(b-2) Pressure differential, 0.052 psia.
(b) Date, 2/6/64; condensing rate, 3.21 pounds per hour.

(c-1) Total pressure, 45.0 psig; reading from Heise gage.
(c-2) Pressure differential, 0.047 psia.
(c) Date, 2/6/64; condensing rate, 0.63 pound per hour.

Figure 10. - Gage pressure traces for fuel-cell condenser.
Figure 11. - Continued. Sequence of movie frames showing slug of condensate emerging from condensing tube.
Figure 11. Continued. Sequence of movie frames showing slug of condensate emerging from condensing tube.
Figure 12. - Experimental determination of bubble pressure in condensing tubes.